



Development and test of a Stirling engine driven by waste gases for the micro-CHP system

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ABSTRACT

In recent years, micro-CHP systems are attracting world attention. As one kind of external heating engines, Stirling engines could be applied to the micro-CHP systems driven by solar, biogas, mid-high temperature waste gases and many other heat sources. The development of a Stirling engine driven by mid-high temperature waste gases is presented first. The thermodynamic design method, the key parameters of the designed Stirling engine and its combustion chamber adapted for waste gases are described in detail. Then the performance test of the Stirling engine is carried out. During the test, the temperature of the heater head is monitored by thermocouples, and the pressure of the working fluid helium in the Stirling engine is monitored by pressure sensors. The relationships among the output shaft power, torque and speed are studied, and the pressure losses of the working fluid in the heat exchanger system are also analyzed. The test results demonstrate that the output shaft power could reach 3476 W at 1248 RPM, which is in good agreement with the predicted value of 3901 W at 1500 RPM. The test results confirm the fact that Stirling engines driven by mid-high temperature waste gases are able to achieve a valuable output power for engineering application.

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1. Introduction

Sharp fluctuations in oil prices and energy supply security risks exist all over the world today. Environmental damage and the effects of atmospheric warming are increasingly serious. In order for sustainable development of economic, resources, environment, the way of producing and using powers should be transformed urgently. In this background, the CHP (Combined Heat and Power) system that can be driven by clean energy such as solar, biogas, exhaust heat and so on has been attracting more and more attention of many countries governments, especially in the European Union, the United States, Canada, Japan and other developed countries.

The engines for CHP systems are the key equipments, such as gas turbines, Diesel engines, gasoline engines, Stirling engines and so on. Different engines should be used for different CHP systems depending on the magnitudes of the required output power. According to different characters of the engines mentioned above and the application area, the heat source could be various, for

example, solar, biogas, waste gases with mid-high temperature. It could be called micro-CHP system if the power supply of the system is less than 5 kW. The Stirling engine, due to its characters of multiple heat source applicability, high thermal efficiency and low noise, is ideal for micro-CHP system, which could be driven by clean energy [1,2]. It is already reported that the international famous Stirling engine that named SOLO/Cleanergy V-161 has been studied for the use into micro-CHP systems which were driven by natural gases [3–5]. Many study work have also been done on the β -type Stirling engines, especially the analytical, simulated, and testing work [6–15]. According to these work, the β -type is proved to be a good structure of the Stirling engine because it could avoid the side force from the piston acting to the cylinder's inner wall by using the rhombic drive mechanism. But so far no experimental study on Stirling engines driven by mid-high temperature waste gases is found from the public literatures.

Exhaust from many industrial equipments are kinds of high-grade heat, such as the waste gases from forging and billet heating furnaces at the temperature between 900 and 1200 °C, and from dry-process cement kilns at the temperature between 600 and 800 °C, and from the glass furnaces at the temperature between 650 and 900 °C [16]. These waste gases can be wisely used as the heat sources to drive CHP systems.

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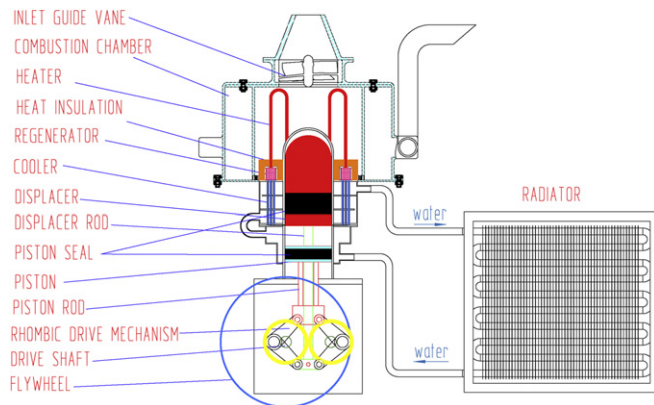


Fig. 1. The structure of the designed Stirling engine driven by waste gases.

We have developed a single-cylinder, β -type Stirling engine prototype with output shaft power of several kilowatts, which could be driven by mid-high temperature waste gases. And the results of our test work confirm the fact. We have also studied the pressure variation of the working fluid in the heat exchanger systems of the Stirling engine during the test process.

Table 1
Some key design parameters of the Stirling engine development by the authors.

		Parameter	Value
Mechanical drive system		Rhombic drive length	72 mm
		Crank radius	26 mm
		Eccentricity	26 mm
		Displacer/piston swept area	81.71 cm ²
		Displacer stroke	5.65 cm
		Piston stroke	4.44 cm
		Expansion cavity swept volume	462.01 mL
		Compression cavity swept volume	362.81 mL
		Phase angle advance	115°
		Number of heater tubes	46
		Heat transfer length of single tube	313 mm (outside) 305 mm (inside)
		Equivalent diameter	27.13 mm
		Designed wall temperature	550 °C
		Total volume	178.62 mL
Heat exchanger system	heater	Structure	Layers of porous metal
		Axial length	40 mm
		Number of wires per inch	200 × 200
		Filler factor	27.04%
	regenerator	Volume for working fluid	217.64 mL
		Equivalent working temperature	204 °C
		Number of cooler tubes	190
		Heat transfer length of single tube	98 mm
	cooler	Equivalent diameter	27.56 mm
		Designed wall temperature	15 °C
		Total volume	58.50 mL
		Working fluid	Helium
	Working state	Designed Speed	1500 RPM
		Mean pressure	2 MPa
		Required heat	15,003 W
		Indicated power output	3901 W
		Thermal efficiency (predicted)	0.26



Fig. 2. The complete Stirling engine with its combustion chamber for waste gases and the radiator.

2. Design of the β -type stirling engine

The single-cylinder, β -type Stirling engine developed by us is designed with the rhombic drive mechanism, which was invented by Philips Company in Netherlands [17,18]. Its main structure and key parts are shown in Fig. 1. The heater was made by 46 U-shape stainless steel tubes, in order to use the heat carried by waste gases wisely.

The thermodynamic design is based on the classic Schmidt analysis method. In order to simplify the design and analysis work, the Schmidt analysis method firstly assumes that the heat source and the heat sink are both isothermal, the working fluid obeys the perfect gas equation, no working fluid leaks, and so on. Then to get the pressure changes of the working fluid in a cycle according to the given key parameters and designed working state, and finally to compute the work done in a cycle by using the first law of thermodynamics. The indicated output power could also be calculated at the designed speed. It is found that the experimental output power are usually no more than 55% of the results calculated by the above Schmidt analysis method [19–21], so we modify the classic Schmidt method by considering the experimental data of the GPU-3 Stirling engine developed by the General Motor Company, which is also a β -type Stirling engine with the rhombic drive mechanism, in order to make the designed/predicted output power of our Stirling engine more reliable than that computed by the classic

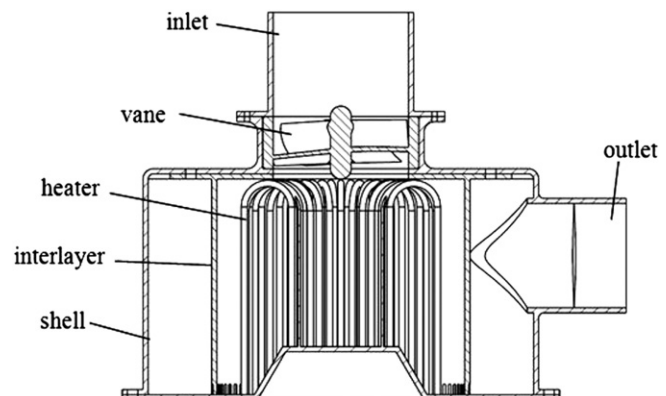


Fig. 3. The structure chart of the designed combustion chamber for waste gases.

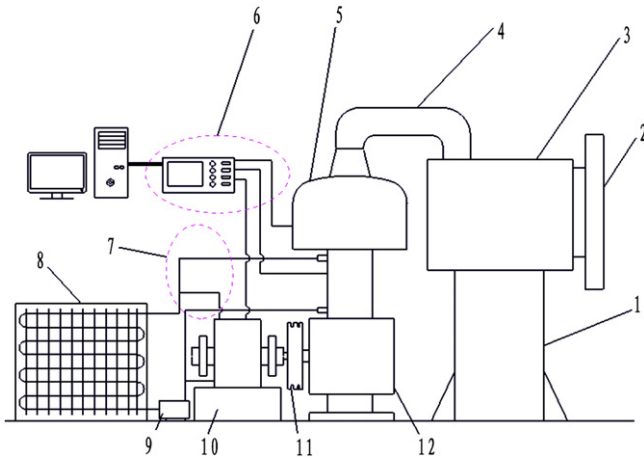


Fig. 4. The layout of the test system for the Stirling engine driven by waste gases. 1-Holder of the gasoline engine; 2 - Radiator for the gasoline engine; 3-Gasoline engine; 4-Bridge for waste gases; 5-Connecting conduit; 6-Test data recorder; 7-Channels of circulating cooling waters; 8-Radiator of cooling waters; 9-Pump of cooling waters; 10-electric eddy current dynamometer; 11-Flywheel of the Stirling engine; 12-Body of the Stirling engine.

Schmidt analysis method. The modification method is to get the ratio of the experimental values to the computed values by the Schmidt analysis method under some working states (the ratio should be a range due to different working states of the GPU-3 Stirling engine under experimental testing), and then to use it as a multiplying factor to the output power computed by the Schmidt method, so the final designed value would agree well with the real output power obtained from the experimental test. Some key design parameters of the Stirling engine are shown in Table 1. The designed/predicted output shaft power is 3901 W, while the required input heat is 15,003 W at the designed working state. Finally, the Stirling engine prototype has been fabricated as shown in Fig. 2.

3. Design of the combustion chamber for waste gases

In the traditional design, the waste gases flow straight-forwardly across the heater of a Stirling engine. The waste gases in the combustion chamber are of low turbulence, and the convection boundary layer is thick, which is disadvantageous for the convective

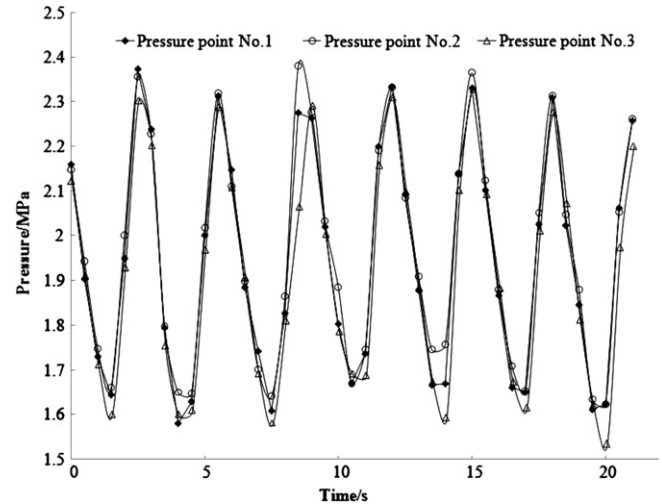


Fig. 6. The test data of the pressure in the heat exchanger system of the Stirling engine.

heat transfer. Neither the thermal efficiency nor the specific output power of the Stirling engine is high.

In view of these disadvantages, we designed a combustion chamber that allows the gases swirling in it, in order to enhance the heat transfer between the waste gases and the heater tubes of the Stirling engine. The structure of this combustion chamber is shown in Fig. 3. The combustion chamber mainly includes a cylindrical shell, a thermal insulating interlayer, an inlet with guide vanes, and a tangential outlet. The mid-high temperature waste gases could obtain a tangential velocity component after flowing through the inlet guide vanes, and then swirling flow is generated in the chamber, thus to intensify the flow turbulence. After releasing heat to the heater tubes, the waste gases flow from the bottom gaps of the interlayer into the outer shell, and then exit the chamber from the tangential outlet. This design could significantly enhance the convective heat transfer between the waste gases and the heater, so as to increase both the specific output power and the engine efficiency due to the high efficiency of retrieving the heat carried by waste gases.

4. Performance test process of the stirling engine

The layout of the test system is shown in Fig. 4. The mid-high temperature exhaust gases from a gasoline engine are used to

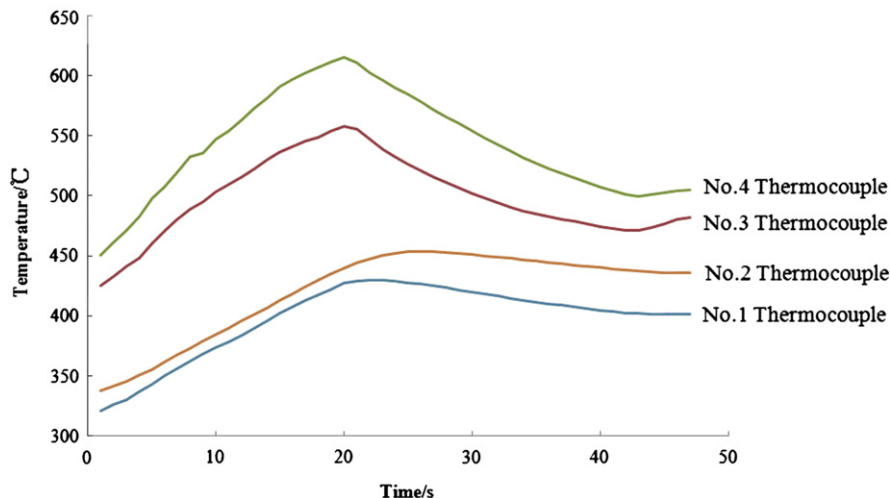


Fig. 5. The test data of the temperature of a heater tube.

Table 2
The average pressure.

Pressure point No.	Average value
1	1.945 MPa
2	1.964 MPa
3	1.919 MPa

drive the Stirling engine in the test. When the gasoline engine is running, the waste gases enters into the combustion chamber through a connecting conduit to release heat to the heater tubes of the Stirling engine, and then flow out from the tangential outlet mentioned above. The output shaft across the center of the flywheel is coupled to an eddy current dynamometer, by which the output power, torque and speed were measured simultaneously during the whole test time. The circulating cooling waters are driven by a pump, in order to cool both the cooler of the Stirling engine and the eddy current dynamometer. A radiator dissipates the heat from the cooling water to the ambient air. In the test, three pressure sensors are used to monitor the pressure of the working fluid within the heat exchanger system of the Stirling engine at three different points. Pressure sensor 1 is located at the joint gap between the cooler and the inlet of the compression chamber. Pressure sensor 2 is located at the joint gap between the cooler and the regenerator. Pressure sensor 3 is located at the joint gap between the regenerator and the heater. Only one heater tube is under temperature monitoring by 4 K-type thermocouples in the test, because all the 46 tubes are arranged axisymmetrically. The required data in the test including temperature, pressure, power, speed, and torque are collected by separate data collection instruments through each corresponding lines. All the data were transferred to the same one computer for displaying and post-processing.

Before operating the test, the indicators of all the collection instruments and the tightness of the Stirling engine are checked out to ensure they were OK. When all is ready, the Stirling engine is made nearly vacuous, and then helium is filled until the pressure sensor averagely reads 2 MPa. Then the valves are closed and the filling process is finished.

The gasoline engine is started at no-load state and the readings of all the 4 thermocouples are paid special attention to. When the average temperature reaches 350 °C, the cooling water pump is opened, and then the starting motor is intermittently operated until the Stirling engine begin to run itself.

When the Stirling engine is running normally, the pressure of working fluid is controlled by the valves, through which helium could flow into or out of the engine. The wall temperature of the heater tubes is controlled by the throttle of the gasoline engine, through which the flow rate and the temperature of the waste gases could be adjusted. These operations are continued until the Stirling engine maintains a relatively stable working state. Data are recorded in all the test time.

When all the test work is finished, the gasoline engine is stopped firstly. After keeping the Stirling engine running for next 2–3 min, the bleed channels are opened until the pressure of the helium inside the Stirling engine dropped to about 1 MPa. Then the deflation is stopped. Finally the circulating cooling waters system is closed after next 20 min, and the test is ended.

5. Test results and discussion

Fig. 5 shows the test data of the wall temperature of the heater tube under monitoring in a period of about 50 s during testing. The temperature captured by No. 4 thermocouple is the highest, which is located near the central axis of the 46 U-shape tubes, while that of No. 1 thermocouple is the lowest, which is located close to the bottom of the heater tube. These results show that the temperature distribution agrees with our expectation in that the tubes at inner side were heated by waste gases firstly while those at the outer were heated subsequently. The combustion chamber functions well. The overall temperature difference is less than 200 °C during the heating processes. In the process of reducing heating, the temperature is almost stable, and the maximum temperature difference is around 100 °C (between No. 1 and No. 4 thermocouple).

Fig. 6 shows the pressure change of the helium in the heat exchanger system when the Stirling engine is working at the state that the mean pressure of it is about 2 MPa in a period of 21 s with a recording time interval of 0.5 s. Each pressure sensor's location is

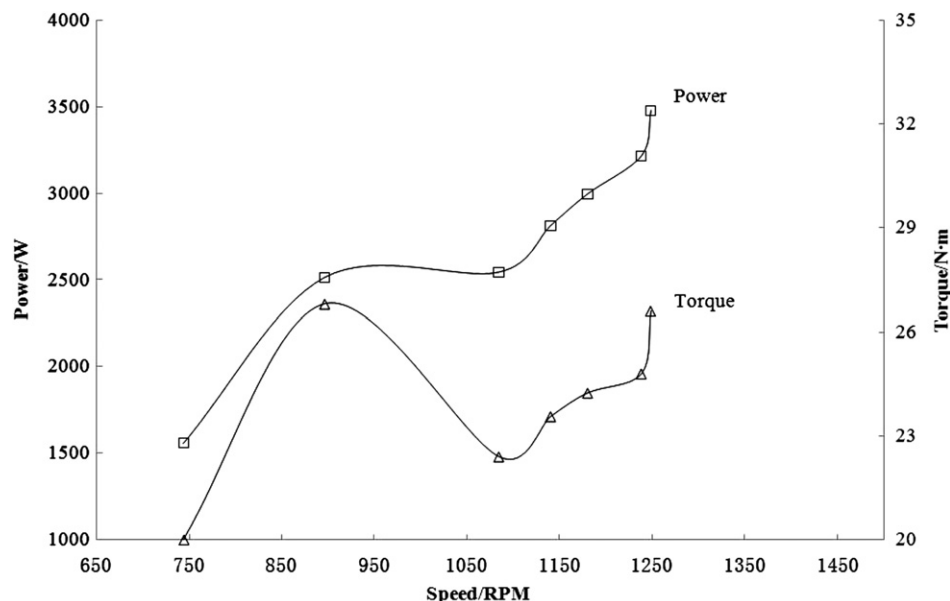


Fig. 7. The test data of the power, torque and speed of the Stirling engine.

mentioned in the section of performance test process of the Stirling engine. As shown in Table 2, the pressure loss of the helium is 0.02 MPa when it flows through the cooler, as observed from the data recorded by the sensor 1 and sensor 2, while the pressure loss is 0.045 MPa when it flows through the regenerator, as observed from the data recorded by the sensor 2 and sensor 3. It also suggests that the pressure difference ratio is 2.3% between the inlet and outlet of the regenerator, which is more than twice of that between the inlet and outlet of the cooler, and the latter is 1%. The results indicate that the porous metal regenerator mainly causes the pressure loss of the working fluid in the heat exchanger system of the Stirling engine. Meanwhile, the shape of the curves shown in Fig. 6 told that the pressure variation processes of the helium are approximately sinusoidal periodic, and the phase difference among the three pressure test points is almost zero.

Fig. 7 shows the relationships among the torque, power and speed of the Stirling engine when it is working. During the test, the maximum torque has reached 26.6 Nm, while the maximum output shaft power has reached 3476 W, and the corresponding speed is 1248 RPM. The results satisfy the corresponding designed/predicted values shown in Table 1, although the speed does not reach the designed value of 1500 RPM. The test data also show that a certain amount of mid-high temperature waste gases could indeed drive some Stirling engines to generate considerable output shaft power to further drive generators or other mechanical machines like pumps. The output power is valuable from a practical point of view. From the upward trend at the end of the curve shown in Fig. 7, the output power and torque have not reached their maximum yet. When the heat carried by waste gases increases, it is expected to output more power and torque.

6. Conclusions

The development and test work of a Stirling engine driven by waste gases for the micro-CHP system is presented in this paper. The maximum output power has reached 3476 W in the test process. And the test work also confirmed the fact that Stirling engines could be indeed driven by mid-high temperature waste gases, and the output power is valuable from a practical point of view.

Comparing with the test results, the classic Schmidt method modified by some experimental data could gain a design value of practical significance, which is close to the reality.

The pressure change of the working fluid in the heat exchanger system of the designed Stirling engine is also studied in the experimental way. The loss between the inlet and outlet of the regenerator is more than twice of that between the inlet and outlet

of the cooler. The porous metal regenerator mainly causes the pressure loss of the working fluid. The pressure variation is approximately sinusoidal periodic, and the phase difference among the three different locations is almost zero.

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